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Analysis on Self-locking of Sliding Vane in a Double Vane Ellipse Rotor Compressor

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(Abstract) The vane motion law and the acting forces on the sliding vane in a double vane ellipse rotor compressor were formulated and analyzed. Based on the analysis of the dynamic characteristic of the sliding vane, self-locking phenomena of the sliding vane was analyzed. A mathematic model for the calculation of the critical parameter of self-locking was established. Calculations on the dynamic characteristic and the critical parameter were performed for a prototype of the double vane ellipse rotor compressor. The calculated results show that the possibility of self-locking increases with the decrease of the length ratio of the shorter and longer axis of the rotor. The pressure ratio is also a factor which affects the possibility of self-locking. Under the condition of normal pressure ratio, self-locking is mainly related to the length ratio of the shorter and longer axis of the rotor.

Keywords: double vane ellipse rotor compressor; sliding vane; self-locking; critical parameter

1. INTRODUCTION

Rotary type compressors were widely used in refrigeration and air-conditioning systems for the advantages of less noise and vibration disturbances. So far, several types of rotary compressors have been designed and developed for the above applications [1-3]. The two typical types of the rotary compressors are the rolling piston compressor and the sliding-vane compressor. Many investigations on the design and performance of the rotary compressor have been carried out [4-8]. The profiles of the rotors in these rotary compressor are always round. The double vane ellipse rotor compressor is one type of the rotary compressor whose rotor is designed as an ellipse.

The profiles of ellipse rotor compressors are ellipses, so the pressure angle between the vane and the rotor is larger than that in a rolling piston compressor. Large pressure angle will lead to larger side force on the vane. Poor loading condition caused by the side force will reduce the life and efficiency of the compressor. A more serious problem is self-locking in this case. It means that the vane can not overcome the resistance at a certain rotation angle no matter how the driving force increases, and then the vane and the compressor both stop operating. So the motion and the forces of the sliding vane in ellipse rotor compressors need the analysis to avoid self-locking.

Taking a single vane ellipse rotor compressor for the example, Guansheng et al. (2006) carried out an analysis on self-locking of the compressor in [9]. But the study ignored the spring force and the gas pressure on both sides and both ends of the vane which have important influences on self-locking, which will lead to a great calculation error of the critical parameter of self-locking. Wei et al. (2012) introduced a double vane ellipse rotor compressor in [10]. Self-locking of the sliding vane is more important to some degree for double vane ellipse rotor compressor.

In this paper, the forces on the vane of the double vane ellipse rotor compressor have been analysed in detail, such as the spring force, the gas pressure on both sides and both ends, the support force and the friction from the sliding-vane slot, etc. Fully considering the details of the vane motion, self-locking in the double vane ellipse rotor compressor has been studied and the critical parameter of self-locking has been obtained. The result provides theoretical foundation for the research design of the double vane ellipse rotor compressor.

2. MOTION ANALYSIS OF THE VANE

Figure 1 shows the motion law of the sliding vane in a double vane ellipse rotor compressor. The upper vane and the lower vane are symmetrically arranged in the cylinder, so their

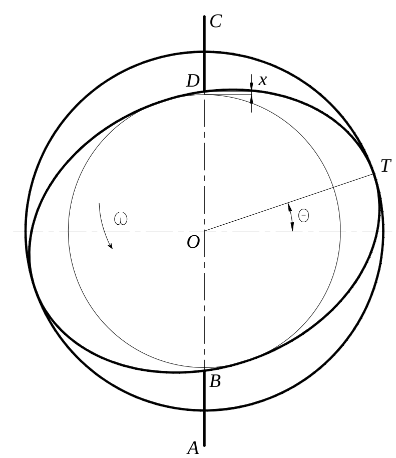

Figure 1. Motion law of the vane.

Nomenclature

a	length of longer half axis [mm]
b	length of shorter half axis [mm]
ε	length ratio of the shorter and longer axis of the rotor
ω	angular velocity [rad/s]
n	rotation speed [r/min]
P_b	suction pressure [Pa]
P_c	discharge pressure [Pa]
H	cylinder height [mm]
m_v	vane weight [kg]
K	spring stiffness [N/mm]
x_0	spring precompression at $\theta = 0$ [mm]
μ_v	friction coefficient between vane and rotor
μ_s	friction coefficient between vane and vane slot

Force situations are only different in the direction, and here the upper vane is taken as the study object. Line CD expresses the upper vane. The vane is a rigid body, so the motion of any point on the vane can represent the motion of the vane itself. Point D represents the contacting point between the vane and the surface of the rotor, which is assumed always on the vertical centerline of the vane. Point T represents the point of tangency between the surface of the rotor and the cylinder. When the rotation angle is $\theta = 0$, the vane inserts its most body into the cylinder, while the displacement of point D, i.e. the lower end of the vane, is zero (here the outward displacement is defined as the positive displacement).

As shown in **Figure 1**, at any rotation angle θ, the displacement of point D can be expressed as:

$$x = \rho - b \tag{1}$$

Here ρ is the length of line OD. It can be expressed as:

$$\rho = \frac{b}{\sqrt{\varepsilon^2 \sin^2\theta + \cos^2\theta}} \tag{2}$$

where

$$\varepsilon = b / a \tag{3}$$

The displacement of the vane can be represented as:

$$x = \frac{b}{\sqrt{\varepsilon^2 \sin^2\theta + \cos^2\theta}} - b \tag{4}$$

By taking the derivative of the displacement given by **Eq.4**, the velocity of the vane can be obtained:

$$v_v = -\frac{1}{2}(\varepsilon^2 - 1) b\omega \sin 2\theta (\varepsilon^2 \sin^2\theta + \cos^2\theta)^{-\frac{3}{2}} \tag{5}$$

where

$$\omega = \pi n / 30 \tag{6}$$

By taking the derivative of the velocity given by **Eq.5**, the acceleration of the vane can also be obtained:

$$\begin{aligned} a_v = {} & (1-\varepsilon^2) b\omega^2 [\cos 2\theta (\varepsilon^2 \sin^2\theta + \cos^2\theta)^{-\frac{3}{2}} \\ & + \frac{3}{4}(1-\varepsilon^2)\sin^2 2\theta (\varepsilon^2 \sin^2\theta + \cos^2\theta)^{-\frac{5}{2}}] \end{aligned} \tag{7}$$

Table 1. operating specifications and the main dimensions of the double vane ellipse rotor compressor.

Length of longer half axis, a	21 mm
Length of shorter half axis, b	17 mm
Cylinder height, H	11 mm
Arc radius of lower end of the vane, r_v	5 mm
Vane length, l_0	10 mm
Vane thickness, B_v	4 mm
Spring stiffness, K	7 N/mm
Spring precompression at $\theta = 0$, x_0	2 mm
Suction pressure, P_b	0.11 MPa
Discharge pressure, P_c	0.8 MPa
Friction coefficient between vane and rotor, μ_v	0.20
Friction coefficient between vane and vane slot, μ_s	0.20
Rotation speed, n	2000 r/min

In this paper, a double vane ellipse rotor compressor is taken as the study object to analyze the self-locking phenomena. The geometric parameters of the prototype are given in **Table 1**.

From **Eq.4**, **Eq.5** and **Eq.7**, the displacement, the velocity and the acceleration of the lower end of the vane within the rotation angle range from 0° to 90° can be obtained, as shown in **Figure 2**, **Figure 3** and **Figure 4**. When $\theta = 0^\circ$, the displacement and the velocity is zero, while the acceleration reaches the positive maximum. When $\theta = 90^\circ$, the displacement reaches the maximum while the acceleration

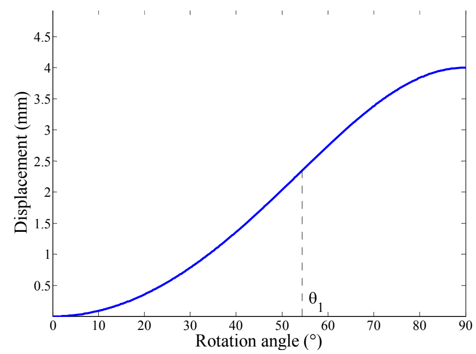

Figure 2. Displacement change caused by rotation angle.

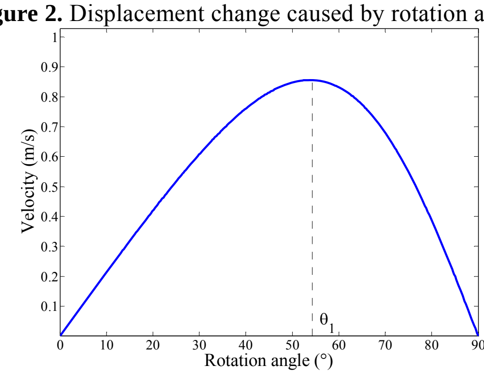

Figure 3. Velocity change caused by rotation angle.

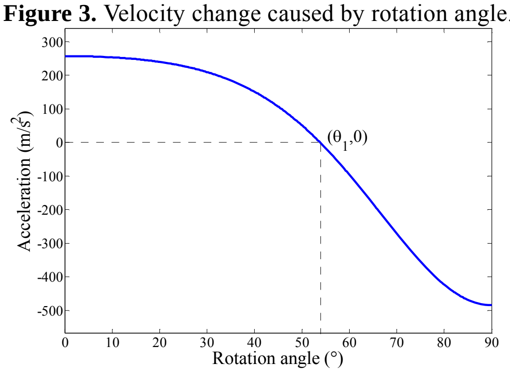

Figure 4. Acceleration change caused by rotation angle.

reaches the reverse maximum. And when $\theta = \theta_1$, the acceleration is zero while the velocity reaches the maximum.

3. FORCE ANALYSIS OF THE VANE

Figure 5 shows the force analysis of the vane. The contact forces between the vane and the rotor are expressed as F_n and F_t respectively. The contact forces between the vane and the sliding-vane slot are F_{R1}, F_{R2}, F_{Rt1} and F_{Rt2}. The spring force is F_k and the inertia force is F_{Iv}. The gas pressure or the lubricating oil pressure has also been shown in **Figure 5**. The arc radius of the lower end of the vane, the length and the thickness of the vane have been expressed as r_v, l_0 and B_v respectively.

As shown in **Figure 6**, the point of tangency between the rotor and the vane is expressed as E. Line O_1E connects point E and the arc center of the lower end of the vane. The angle between O_1E and the centerline of the vane is α. The distance between point E and the centerline of the vane is d. Line OE connects point E and the geometric center of the rotor. The angle between OE and the longer axis is φ.

From **Figure 6**, the slop of the tangent on the point of tangency between the rotor and the vane can be written as:

$$y' = \tan\alpha = \tan(\arctan(-\frac{a^2}{b^2}\cot\varphi) + \theta) \tag{8}$$

The distance between point E and the centerline of the vane can be written as:

$$d = r_v \sin\alpha = \frac{b}{\sqrt{\varepsilon^2\cos^2\varphi + \sin^2\varphi}}\sin(\frac{\pi}{2} - \theta - \varphi) \tag{9}$$

From **Eq.8** and **Eq.9**, we obtain α:

$$\alpha \approx \theta - \arctan\left\{\varepsilon \cot \arctan\left[\frac{1}{\varepsilon}\tan(\frac{\pi}{2} - \theta)\right]\right\} \tag{10}$$

In order to simply the calculation, the pressure in the clearances of both sides of the vane can similarly be considered to offset each other. As shown in **Figure 5**, the pressure difference between the two ends of the vane can be represented as:

$$F_c = H[B_v P_0 - P_b(\frac{B_v}{2} + r_v \sin\alpha) - P_c(\frac{B_v}{2} - r_v \sin\alpha)] \tag{11}$$

where P_0 is the pressure on the backside of the vane and it equals P_c commonly.

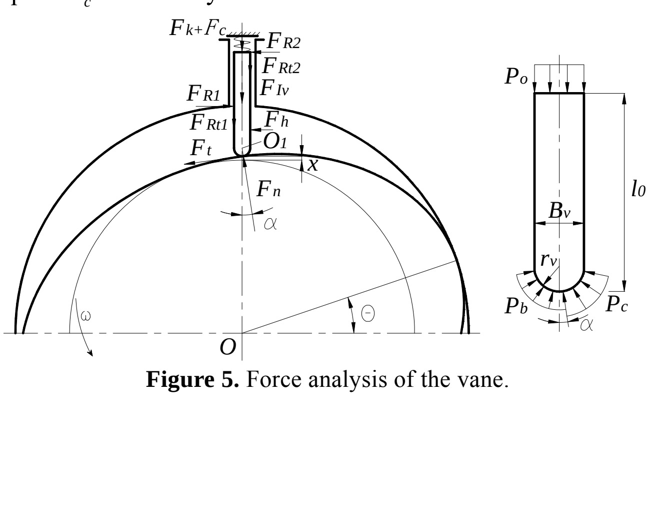

Figure 5. Force analysis of the vane.

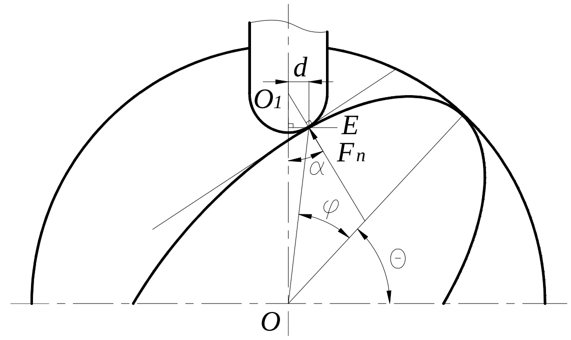

Figure 6. Geometrical relationship at the point of tangency between the rotor and the vane.

The pressure difference between the two sides of the vane inside the the cylinder can be expressed as:

$$F_h = H(a - b - x)(P_c - P_b) \tag{12}$$

The inertia force on the vane can also be obtained:

$$F_{Iv} = -m_v a_v \tag{13}$$

The spring force on the vane can be represented as:

$$F_k = K(x_0 + x) \tag{14}$$

The friction and the positive contact force between the vane and the rotor have the form:

$$F_t = \mu_v F_n \tag{15}$$

The friction and the positive contact force between the vane and the sliding-vane slot takes the form:

$$F_{Rt1} = \mu_s F_{R1} \tag{16}$$

$$F_{Rt2} = \mu_s F_{R2} \tag{17}$$

The force equilibrium equations of the vane are as follows:

$$F_n\cos\alpha - F_t\sin\alpha - F_{Iv} - F_{Rt1} - F_{Rt2} - F_k - F_c = 0 \tag{18}$$

$$F_{R1} - F_{R2} - F_h - F_n\sin\alpha - F_t\cos\alpha = 0 \tag{19}$$

Taking the arc center of the lower end of the vane as moment center, the moment equilibrium equation can be expressed as:

$$\begin{aligned}(F_{Rt1} - F_{Rt2})\frac{B_v}{2} + F_{R2}(l_0 - r_v) - F_{R1}(a - \rho \\ - r_v) - F_t r_v - F_h\frac{(r_v\cos\alpha)^2 - (a - \rho - r_v)^2}{2(a - \rho - r_v + r_v\cos\alpha)} = 0\end{aligned} \tag{20}$$

The moment equilibrium equation, taking the point of tangency between the rotor and the vane as moment center, can be represented as:

$$\begin{aligned}F_{R2}(l_0 - r_v + r_v\cos\alpha) + F_h\frac{a - \rho}{2} + F_{Rt1}(\frac{B_v}{2} + r_v\sin\alpha) \\ +(F_k + F_c)r_v\sin\alpha - F_{R1}(a - \rho) - F_{Rt2}(\frac{B_v}{2} - r_v\sin\alpha) = 0\end{aligned} \tag{21}$$

From **Eq.18**, **Eq.19**, **Eq.20** and **Eq.21**, the inertia force on the vane can be obtained. Then the resultant force on the vane is obtained:

$$F = -F_{Iv} \tag{22}$$

4. SELF-LOCKING ANALYSIS OF THE VANE

As shown in **Figure 4**, the acceleration of the vane is greater than zero within $(0, \theta_1)$, while the resultant force on the vane is always upward. When $\theta = \theta_1$, the velocity reaches the maximum, while $a_v = 0$. The acceleration of the vane is less

than zero within $\left(\theta_1, 90^\circ\right)$, while the resultant force on the vane is always downward.

The upward resultant force leads to the increase of the velocity of the vane within $(0, \theta_1)$, and the velocity increases to the maximum at θ_1. Then the downward resultant force results in the decrease of the velocity of the vane within $\left(\theta_1, 90^\circ\right)$, and the velocity decreases to zero at 90° where the vane reaches the top dead center. However, if the downward resultant force has stopped the vane before it reaches the top dead center, i.e. the velocity has been decreased to zero while $\theta < 90^\circ$, the self-locking phenomena will occur.

For double vane ellipse rotor compressor, F_n is the driving force which makes the rising of the vane. However, the increase of F_n will lead to the increase of F_{Rt1} and F_{Rt2} at the same time. When the self-locking has occurred, the component in the motion direction of the driving force is always equal to the composition of the static friction, the spring force and the gas pressure, which results in that the vane can not continue to move.

Analyzing from the aspect of the energy conservation, if all the kinetic energy generated by the upward resultant force has been converted to potential energy by the downward resultant force before reaching the top dead center, the kinetic energy will be reduced to zero before 90°. At this point, the resultant force on the vane is zero, so no kinetic energy will be generated, which leads to the static state of the vane, then the self-locking occurs.

If the self-locking occurs at θ_2, we obtain:

$$\bar{F}' \cdot S' = 0 \tag{23}$$

where $\bar{F}'$ is the average resultant force of the vane within $(0, \theta_2)$, and S' is the moving distance of the vane within $(0, \theta_2)$.

As shown in **Figure 4**, the acceleration decreases gradually within $\left(0, 90^\circ\right)$. Upon **Eq.23**, the energy equation of self-locking takes the form:

$$W = \bar{F} \cdot S < 0 \tag{24}$$

where W is the surplus kinetic energy at 90°, $\bar{F}$ is the average resultant force of the vane within $\left(0, 90^\circ\right)$, and S is the moving distance of the vane within $\left(0, 90^\circ\right)$.

From **Eq.24**, it has been kown that if $W<0$ at $\theta = 90^\circ$, the vane will self-lock theoretically.

Eq.24 can be represented as:

$$\int_0^{\frac{\pi}{2}} F dx < 0 \tag{25}$$

where

$$dx = -\frac{1}{2}(\varepsilon^2 - 1) b \sin 2\theta (\varepsilon^2 \sin^2\theta + \cos^2\theta)^{-\frac{3}{2}} d\theta \tag{26}$$

5. RESULT AND ANALYSIS

The operating specifications and the main dimensions of the double vane ellipse rotor compressor prototype have been given in **Table 1**. The self-locking phenomena of the vane is analysed based on the parameters shown in **Table 1**. From **Eq.24**, **Eq.25**, **Eq.26**, the surplus kinetic energy of the vane can be obtained. Calculating the surplus kinetic energy with the length ratio of the shorter and longer axis of the rotor and the pressure ratio as the independent variables, we can obtain the changing curves of the surplus kinetic energy as shown in **Figure 7** and **Figure 8**.

Let the longer half axis always be 21mm, the discharge pressure always be 0.8MPa, and the suction pressure always be 0.11MPa, then the pressure ratio will be a fixed value. Taking the length ratio of the shorter and longer axis as the independent variable, we can get the curve of surplus kinetic energy change caused by the length ratio of the shorter and longer axis, as shown in **Figure 7**. When the shorter half axis is greater than or equal to 14.1mm, the surplus kinetic energy is greater than zero, that means the self-locking will not occur. However, if the shorter half axis is less than 14.1mm, while the length ratio of the shorter and longer axis is 0.6741, the self-locking will occur. From the trend of the curve in **Figure 7**, the greater the the length ratio of the shorter and longer axis is, the more difficult the self-locking occurs. But the increase of the length ratio of the shorter and longer axis may lead to the decrease of the efficiency of the compressor. So in the design of the double vane ellipse rotor compressor, the length ratio of the shorter and longer axis should be greater than the critical parameter of self-locking, but which should be under the premise that the efficiency and the performance will not be affected.

Let the suction pressure always be 0.11MPa, the longer half axis always be 21mm, and take the pressure ratio as the independent variable, then we can get the curves of surplus kinetic energy change caused by the pressure ratio under different length ratios of the shorter and longer axis, as shown in **Figure 8**. When the shorter half axis is greater than or equal to 15mm, the surplus kinetic energy is greater than zero within the pressure ratio range from 0.6 Mpa to 1.0 Mpa. However,when the shorter half axis is less than or equal to 14mm, the self-locking will occur within the pressure ratio range. It can also be observed that when the length ratio of the

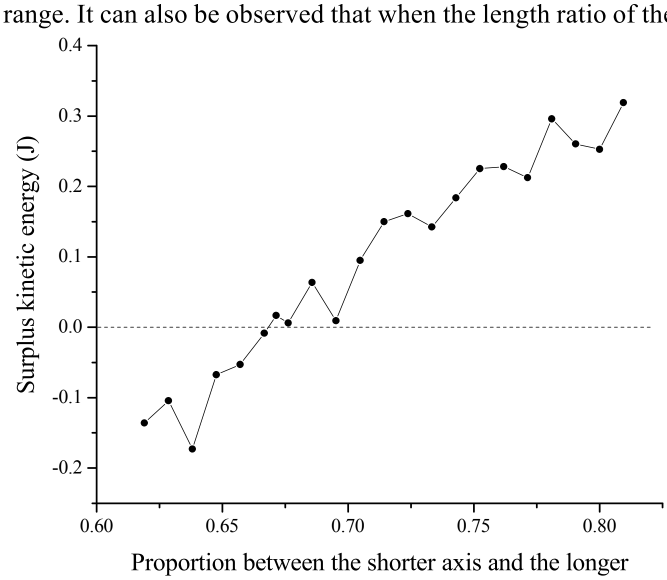

Figure 7. Surplus kinetic energy change caused by proportion between the shorter axis and the longer.

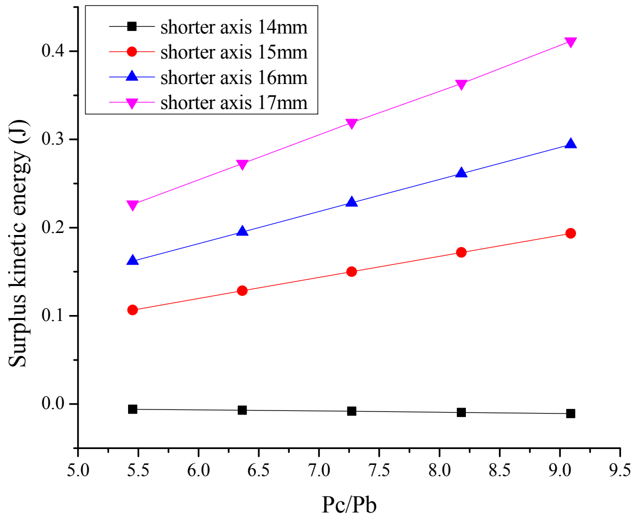

Figure 8. Surplus kinetic energy change caused by pressure ratio.

Shorter and longer axis is greater than the critical parameter of self-locking, the surplus kinetic energy increases with the increase of the pressure ratio, and self-locking is more difficult to occur. On the contrary, when the length ratio of the shorter and longer axis is less than the critical parameter, the greater the pressure ratio is, the easier the self-locking occurs. So it is kown that the pressure ratio is not the critical factor affects self-locking, instead the length ratio of the shorter and longer axis affects self-locking greatly. In the design of the double vane ellipse rotor compressor, the pressure ratio has a wide range of choice.

6. CONCLUSION

In this paper, the motion law and the forces of the sliding vane in a double vane ellipse rotor compressor were theoretically analysed. Further more, self-locking phenomena which occurs to the sliding vane has been analyzed based on the analysis of the dynamic characteristic of the sliding vane. The critical parameter of self-locking has been obtained through the calculation. The length ratio of the shorter and longer axis of the rotor is the major factor in self-locking. The greater the ratio is, the closer the ellipse to be round, the smaller the possibility of self-locking is. The pressure ratio is another factor which affects the possibility of self-locking. Under the condition of normal pressure ratio, the possibility of self-locking is mainly related to the length ratio of the shorter and longer axis of the rotor.

It can be concluded that the length ratio of the shorter and longer axis of the rotor is a major parameter that should be considered in the design of the double vane ellipse rotor compressor. It does not only affect the self-locking, but also influence the volumetric displacement and the capacity utilization. The present result provides the theory basis and the reference opinion for the structural design of the ellipse rotor compressor.

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Author Introduction

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